AN INVESTIGATION OF SHIP MODEL MANEUVERS FOR USE WITH SYSTEMS IDENTIFICATION

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Submitted to the Department of Ocean Engineering in May, 1977, in partial fulfillment of the requirements for the degree of Ocean Engineer and the degree of Master of Science in Naval Architecture and Marine Engineering.

ABSTRACT

This thesis was a second attempt to produce actual data from ship model tests for use with the systems identification process.

The hull form used in these tests was a 1:84 scale model of a destroyer. The model was outfitted with twin rudders, propellers, and a D.C. electric motor for self-propulsion. In addition, instruments were installed to measure rudder angle, sway acceleration, yaw velocity, and roll velocity.

Tests were conducted with the model moving down the tank at a constant forward velocity, while the amplitude of the rudder deflection was varied sinusoidally. Model motions were kept as close to the linear region of motion as possible.

It was necessary to electronically filter the data prior to recording in order to eliminate the roll response at model natural frequency, which was easily excited by the wires that electrically connected the model with power sources and recording instruments. Another source of extraneous signals resulted from machinery vibrations. Bandpass filters were used to filter out all signals but those in the same frequency range as the rudder oscillations.

Though the results of this experiment were inconclusive, several important considerations in conducting further model tests of this type were pointed out. Specific areas needing improvement prior to conducting further tests include the design of an integrator to integrate the roll velocity signal from the rate gyroscope output, design of a lighter weight propulsion system, and reduction of the effect on the model motions by wires that electrically connect the model to power sources and recording instruments.

Thesis Supervisor: Martin A. Abkowitz

Title: Professor of Ocean Engineering

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In attempting to complete this work, it was encouraging to find that many people took time from their busy schedules and demonstrated a great deal of genuine interest in helping me solve problems.

Sam, in the Mechanical Engineering Department Student Workshop, took the time to instruct me in the use of milling machines and lathes in order to construct the mechanical components for the scotch yoke, the twin rudders, and various equipment mountings and shaft supports. Without this assistance, it would have not been possible to outfit the model.

Mr. Ysabel Mejia, the towing tank technician, supplied me with a great deal of encouragement at times when it appeared to me that my problems were insurmountable. He demonstrated a great deal of patience in helping me test the various electrical circuitry necessary to make measurements.

Mr. Ed McCaffrey, from the Department of Civil Engineering, provided technical assistance and advice concerning the electrical circuitry and provided a power supply.

Finally, I would like to express my sincere appreciation to Professor Martin A. Abkowitz, my thesis advisor. His forebearance and patience in dealing with me were indeed

exceptional. Countless times, when I felt as though the situation was hopeless, he provided advice, encouragement, and allowed me to complete the work even though various administrative deadlines were stretched. I am sincerely grateful for his understanding and flexibility.

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CHAPTER 1

INTRODUCTION

This thesis was a second attempt to make an analysis of the results of using the Kalman statistical filter technique of systems identification with real data generated from free running ship model tests at the MIT towing tank. Reference 4 describes the application of the Kalman filtering technique to ship maneuvering analysis using simulated full scale ship data from a computer model. Lee's work as described in Reference 3 was a first analysis of the system identification procedure with real data generated from unconstrained ship model tests.

What was attempted here was to incorporate into the model tests some of the recommendations made in Reference 3. In that work it was discovered that measurements of the sway acceleration of the ship model had superimposed on them a gravitational acceleration component that was proportional to the sine of the roll angle. Even for angles of roll as small as one degree the gravitational component was of the same order of magnitude as the sway acceleration. In the tests performed for this thesis the roll angle was measured during the tests and the gravitational component of sway acceleration removed.

Primary emphasis in this work was an analysis of the results of using the systems identification technique with real data. Lundblad, in Reference 4 after using the systems identification technique with computer-simulated data pointed out that a sinusoidal rudder deflection maneuver gave results that compared favorably with actual values of the hydrodynamic coefficients. So in this work a sinusoidal maneuver was duplicated, and no attempt was made to analyze some range of types of maneuvers over which the technique of systems identification was best suited. Only two of the state variables, yaw velocity and sway acceleration, were measured independently in the tests for this thesis. The coefficients selected for identification were Y_v, $N_r - mx_G u_o$, N_v, $Y_r - mu_o$, and N_δ (or $1/\delta$). The coefficients are used in the criteria for straightline stability, $Y_v(N_r - mx_G u_o) - N_v(Y_r - mu_o) > 0$ from References 1 and 2.

CHAPTER 2

FULL SIZE SHIP DESCRIPTION

The hull form utilized in the tests was a 1:84 scale model of a destroyer type ship. Table 1 lists the various full size hull form characteristics.

Table 1
Full Size Ship
Hull Form Characteristics

Length of Waterline	525 feet
Beam	54 feet
Draft	17.4 feet
Prismatic Coefficient	0.61
Displacement	6,800 tons
Wetted Surface Area	30,600 feet^2

A U.S. Navy type AN/SQS - 26CX bow mounted sonar dome at 21 inches aft of the forward perpendicular on the ship was also molded into the fiberglass ship model used in the tests. Figure 1 is a body plan of the full size ship.

Propellers and rudders used during these tests and the locations of those appendages were scaled down from the U.S. Navy destroyer of the DD-963, Spruance class. However, the hull form used in the tests was not a geosim of the Spruance class destroyer which has a smaller prismatic coefficient.

FIGURE 1

SHIP BODY PLAN

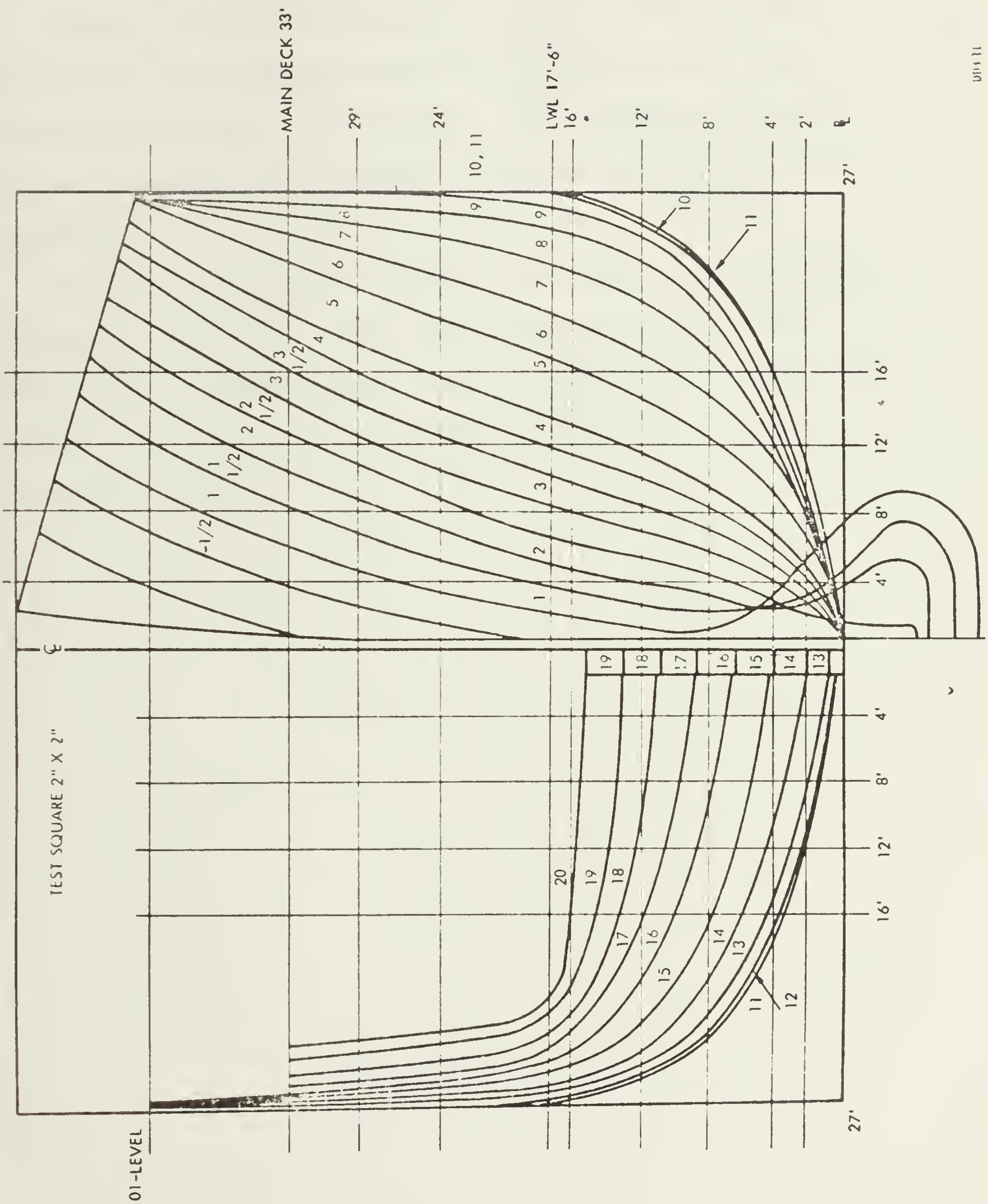


Table 2 lists the various pertinent propeller and propeller shaft characteristics for the full size ship. Propeller rotation was outboard. The starboard propeller rotated clockwise and the port propeller rotated counter-clockwise.

Characteristics for the full size ship's twin rudders are shown in Table 3. The root chord was of the type NACA 0021 wing and the tip chord of the type NACA 0010 wing section. The rudder stock location was at 22 percent of the leading edge.

Table 2

Propeller and Propeller Shaft Characteristics
Full Size Ship

Number of Propellers	2
Propeller Diameter	17 ft.
Number of Blades	5
Design Pitch (ρ/D) at 0.7R	26.18 ft.
Expanded Area Ratio	0.75R
Propeller Shaft Diameter	26.75 in.
Propeller Shaft Location	12 ft. - 9 in. off ship centerline
Propeller Location	17 ft. forward of rudder centerline
Tip Submergence	8 ft.
Tip Clearance	5 ft. 7 in.

Table 3

Rudder Characteristics
Full Size Ship

Root Chord	12 ft. - 6 in.
Tip Chord	7 ft. - 6 in.
Mean Span	14 ft. - 0 in.
Mean Chord	10 ft. - 0 in.
Geometric Aspect Ratio	1.4
Taper Ratio	0.60
Root Thickness	2 ft. - 8 13/16 in.
Tip Thickness	0 ft. - 9 in.
Profile Area	140 $ft.^2$
Sweep Angle of 1/4 Chord	4.3 degrees

CHAPTER 3

CONSTRUCTION OF THE MODEL

A model hull for the tests was already available at MIT. The fiberglass hull was a 1:84 scale model of the DD-963 type U.S. Naval Destroyer described in the proceding chapter. The length of the waterline on the model is 75 inches and the beam is 7 1/2 inches. It was necessary to outfit the model with a means of self-propulsion and maneuvering, and with instrumentation. Table 4 lists the various scaled down characteristics for the ship model.

3.1 Maneuvering System

There were three constraints on the maneuvering system for the ship model. Since the tests were to estimate hydrodynamic coefficients from linearized equations of motion, it was necessary to ensure that ship model motions remained in the linear region. The rate of rudder motion and maximum allowable rudder deflections were the two other constraints.

Prevention of the excitation of significant non-linear motions of the ship model required that the rudder deflection be ± 10 degrees or less. Rudder motion in the ship model was provided by the "scotch yoke" mechanism used in the

Table 4

Ship Model Characteristics

Scale Ratio	1:84
LWL	75 inches
Beam	7 1/2 inches
Displacement	25.70 lbs.
Propeller Diameter	2 27/64 inches
Shaft Location	1 13/16 off centerline
Number of Shafts	2
Propeller Location	2 27/64 inches forward of rudder centerline
Tip Submergence	1 9/64 inches
Tip Clearance	51/64 inches
Number of Propellers	2
Number of Rudders	2
Rudder Location	1 3/4 inches forward of after perpendicular 1 19/32 inches off centerline
Rudder Rate	3.5 degrees in 0.109 sec
Rudder Area	2.85 in.^{2}

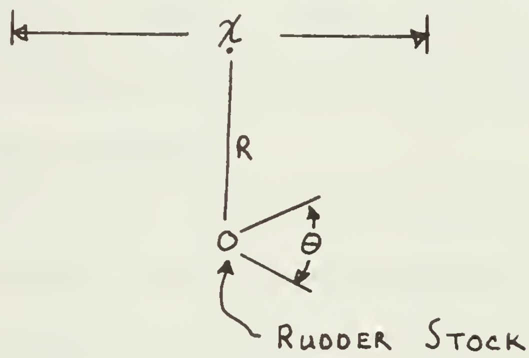
tests described in Reference 3. The equation that governs the amount of rudder deflection is

$$x = 2R \sin \theta/2.$$

x = linear motion of scotch yoke

R = filler length

θ = rudder deflection



The linear motion x is limited by the scotch yoke, $0.823 \text{ in} \leq x \leq 1.594 \text{ in}$. Filler length of 2 5/8 inches was selected based on a rudder deflection of $\theta = 20$ degrees and physical limitations within the ship model.

Since the full size ship rudder rate is 3.5 degrees per second, this scaled down to 3.5 degrees in 0.109 seconds for the ship model. The scaled down rudder rate could be achieved if the scotch yoke speed was adjusted to 5.346 RPM. That speed was within the speed range of

the scotch yoke which could be adjusted from 0 RMP to 18.25 RPM with a variable DC power supply.

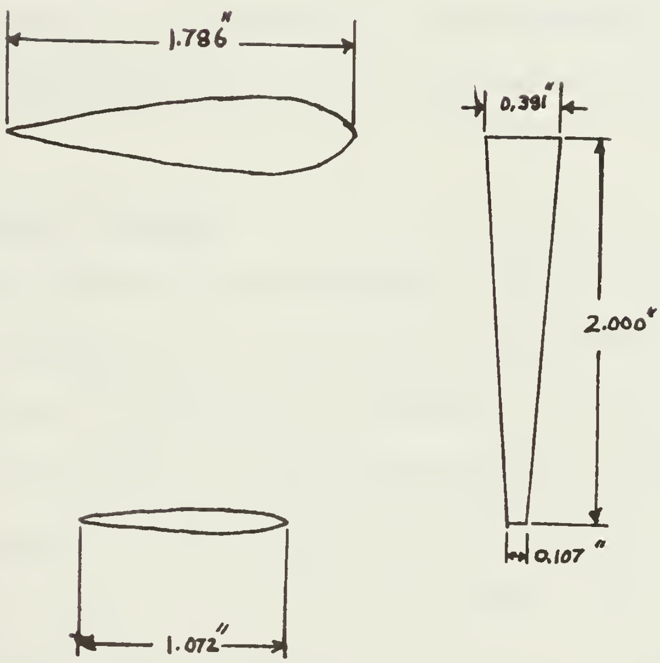
The maximum rudder deflection for the full size ship is ± 35 degrees. During the model tests it was necessary to ensure that those limits were not exceeded.

The twin rudders for the ship model were manufactured in the MIT student workshop. The rudder profiles were milled from one-half inch aluminum plate and the foils shaped with a file. Actual rudder dimensions are shown in Figure 2. Stainless steel 3/16 inch rudder stocks were threaded into the rudders.

An aluminum plate was fixed into the bottom of the ship model to form a flat, level surface through which the rudder stocks passed. The plate, prior to being fixed into the ship model, was drilled for the proper relative locations of the rudder stocks. The plate was secured to a flat epoxy surface in the bottom of the ship model. Holes were drilled through the bottom of the ship model using the plate as a template. Brass tubing was used for a bushing in the holes in the bottom of the ship model. The tubing was secured in place with silicon rubber. The upper ends of the rudder stocks were fixed in place by an aluminum bracket that extended athrawrtships. The rudder stocks

FIGURE 2

RUDDER FOR SHIP MODEL



were held in place by brass bushings. Stainless steel collars held the rudders in a fixed vertical position.

Since it was necessary to move two rudders simultaneously, a new mechanism to transmit the linear motion of the scotch yoke had to be manufactured in the student workshop. Drawings of that device are shown in Figure 3.

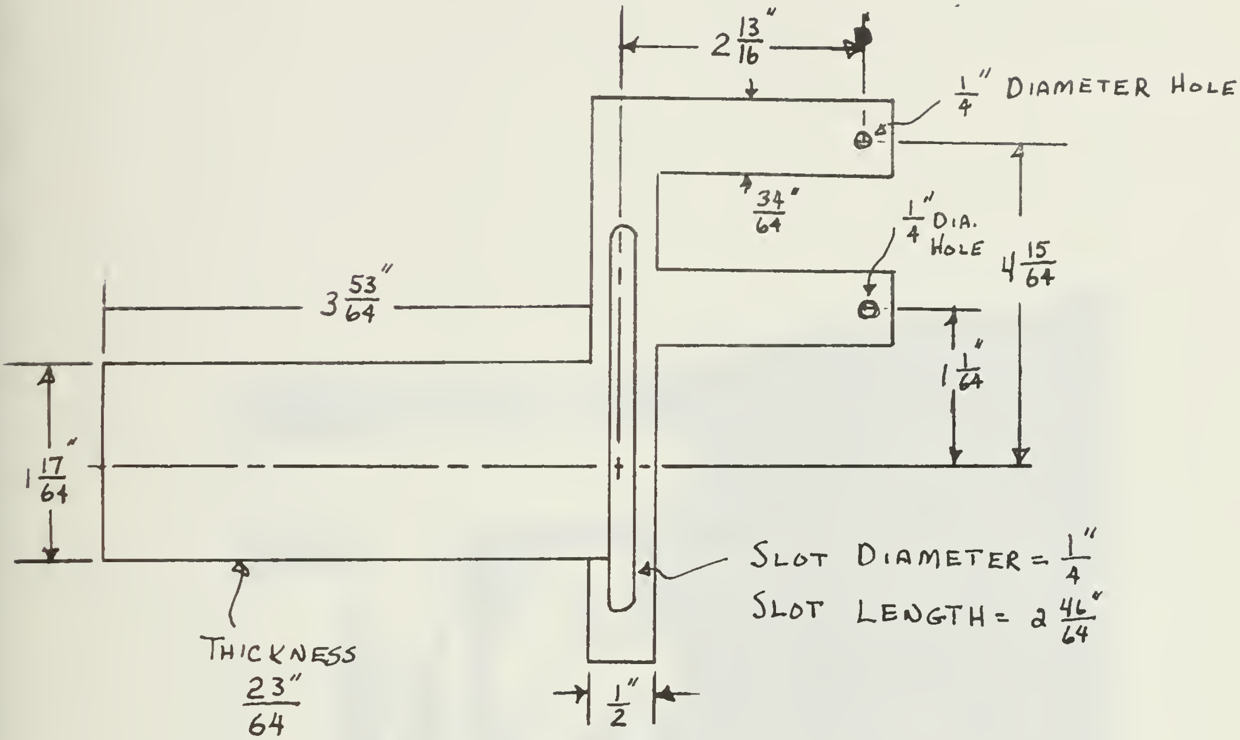
Arrangements of the scotch yoke mechanism and the completed maneuvering system are shown in Figure 4.

3.2 Propulsion System

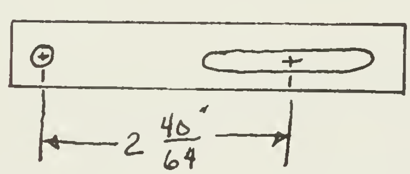
Construction and installation of the ship model propulsion system with twin shafts and counter-rotating propellers was a challenging problem. Finding commercially-available right-hand and left-hand propellers was the first task undertaken. The propellers on the full size ship are five-bladed, variable pitch propellers. In these tests the hull forces were dependent on the jet thrust over a given disc area. Therefore, only the disc area of the propellers was scaled. Right-hand and left-hand rotation propellers were pruchased from Bliss Marine James in Dedham, Massachusetts. The propellers were made of brass with three blades, and were 2 1/2 inches in diameter with a pitch of 7 inches. The propeller size for the model as

Figure 3

Linear Motion Transmission Mechanism and Tiller



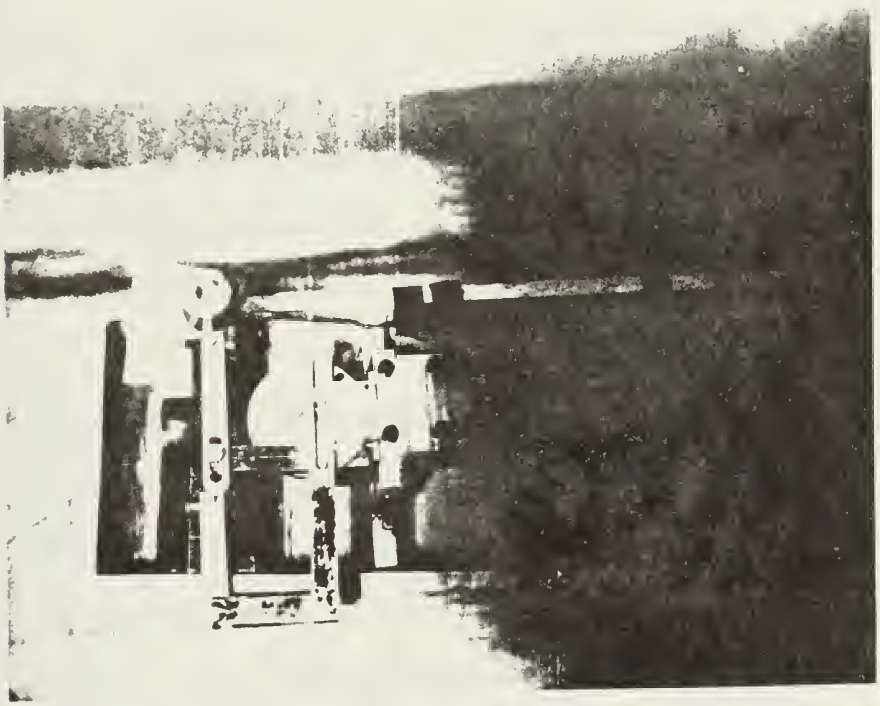
Linear Motion Transmission Mechanism



Tiller

FIGURE 4

MANEUVERING SYSTEM



scaled down from the full size ship should have been 2.43 inches. A 3 percent error in the propeller diameter was considered acceptable for these tests.

Two 3/16 inch drill rods were used for the propeller shafts. Shaft supports and bearings were provided by a single strut, a stuffing tube with needle bearings, and a bushing located just aft of amidships. Figures 5 and 6 show the complete shaft arrangement.

The propulsion motor was a 1/20 horsepower DC electric motor, which was bolted to a flat aluminum plate in the hull.

A gearbelt and pullup were used to drive the port shaft directly from the electric motor. The port shaft was then geared to the starboard shaft. The gears were identical gears with a pitch diameter of 3.75 inches, which meant the shaft center - to center distance was 3.75 inches. The scaled down center-to-center distance was 3.64 inches. Again the 3 percent error was considered acceptable for the tests.

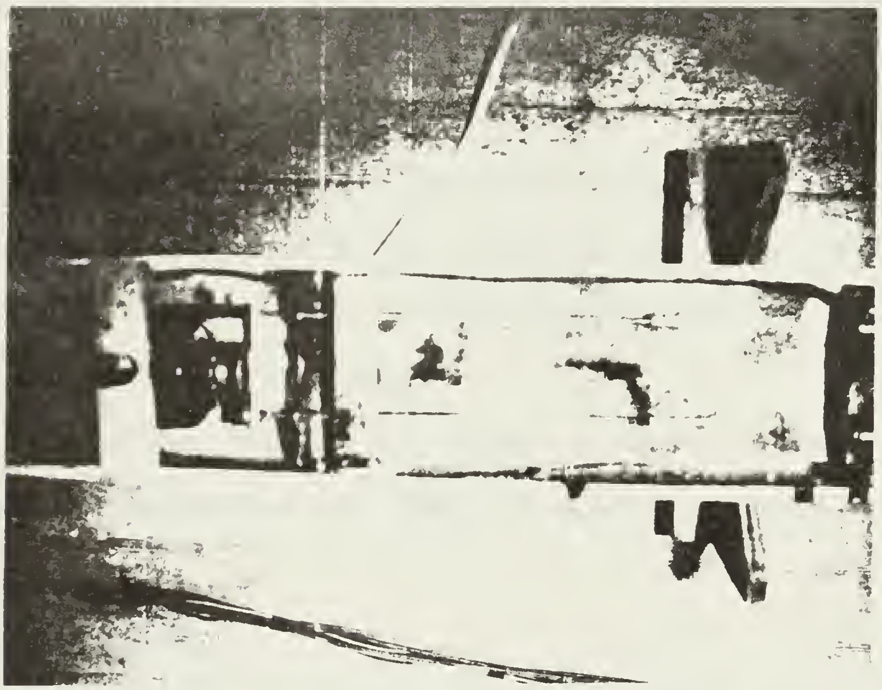
Once a means of supporting the propulsion shafting was devised it was necessary to cut hull penetrations for the shafts. The support bracket at the forward end of the shafts near amidships was installed and bolted to the hull.

FIGURE 5

PROPELLER AND SHAFT
ARRANGEMENT

FIGURE 6

INTERNAL SHAFTING



A bracket was bolted to the hull near station 14; holes were cut in the hull through which the after struts passed and were secured to that bracket. With two brackets and shaft supports in place two holes were cut in the hull near station 15 for the shaft penetrations. When the shafts were in place, stuffing tubes with needle bearings were placed on the shafts and pushed along each shaft passing through the holes in the fiberglass hull. The holes in the hull were patched with fiberglass and epoxy. The fiberglass patch was molded onto the stuffing tube fixing the tube in place. Small holes in the hull around the struts were sealed with silicon rubber.

The underwater hull in the area of shaft penetrations was sanded and repainted to obtain a smooth finish.

3.3 Electric Power

Separate variable DC power supplies were required for the propulsion motor and the scotch yoke mechanism. Power supplying apparatus was external to the ship model and was connected to the ship model through a wiring arrangement described later in this work.

CHAPTER 4

THE MODEL TESTS

4.1 Ballasting the Model

An important consideration in the tests is to insure that the model is properly scaled with respect to the weight distribution about the yaw axis. For an elongated body such as a ship form, the weight distribution about the pitch axis approaches that about the yaw axis. The standard procedure for ballasting ship models at the MIT Towing Tank, Appendix 3, was utilized to ballast the model. From this procedure, it was determined that the heave frequency in air, f_H, is 0.7 cycles per second and the pitch frequency, f_P, is 0.4 cycles per second.

Actual model weight with ballast, propulsion system, maneuvering system, and instrumentation exceeded the scaled model weight of 26.5 pounds. The scaled weight was exceeded prior to ballasting because of the heavy components of the maneuvering system and propulsion system. The model was trimmed by the stern prior to ballasting. To minimize the excess weight, metal filings were used to trim the model to an even keel. A cloth bag was fitted into the ship model's sonar dome, and the filings were added until the model floated without trim. The bag was closed with a string and secured in place with clay. Some additional

ballast was added amidships to correct a slight port list as a result of the propulsion motor being off the model centerline. The final model weight was 30.6 pounds. The natural roll frequency of the model is 1.11 cycles per second which scales up to a natural roll frequency of about 10 cycles per second for the ship. Since the scaled-up roll frequency is about what is expected for a destroyer, the metacentric height was properly scaled on the model.

4.2 Instrumentation

To use the systems identification procedure to evaluate the hydrodynamic coefficients, measurements of the linear forward velocity, rudder angle, sway acceleration, and yaw velocity must be recorded. The sway acceleration is measured by an accelerometer. However, the accelerometer does not measure a pure sway acceleration. The measurement consists of three components:

$$\text{Accelerometer Measurement} = \dot{v} + rU = g \sin\phi$$

$\dot{v}$ = sway acceleration

r = yaw rate

U = linear forward velocity

g = acceleration of gravity

ϕ = roll angle

The rU component is a centripetal acceleration that results from the use of a moving axis system. The $g \sin\phi$ term arises because when the model rolls the accelerometer is no longer being oriented in a horizontal plane. In Reference 3 it was discovered that the term due to the roll of the model is of the same order of magnitude as actual sway acceleration so it is necessary to measure roll angle and correct the accelerometer measurements for roll angle. Therefore, in addition to the measurement quantities mentioned previously, it is necessary to record the roll angle of the model rather accurately.

A three-axis rate gyroscope was used to measure the rate of roll and the rate of yaw. The rate of roll can then be integrated to obtain the roll angle.

To calibrate the rate gyroscope the scotch yoke mechanism was removed from the model and mounted on a clamp. The speed of the scotch yoke turntable was adjusted to 20 degrees per second with the rate gyro mounted on the scotch yoke turntable. The speed of the scotch yoke was checked by using a stop watch to measure the time of one complete revolution of the turntable. Once the scotch yoke was properly adjusted for 20 degrees per second, the rate gyroscope was connected to its power supply and the signal from the axis of the gyroscope to be calibrated was fed

back to a strip chart recorder in the control room for calibration. Each axis of the gyroscope was calibrated separately. The wires leading to the rate gyroscope twisted as the scotch yoke turntable rotated. However, the speed of rotation was slow and a good signal from the gyro can be recorded before the twist of the wires becomes significant.

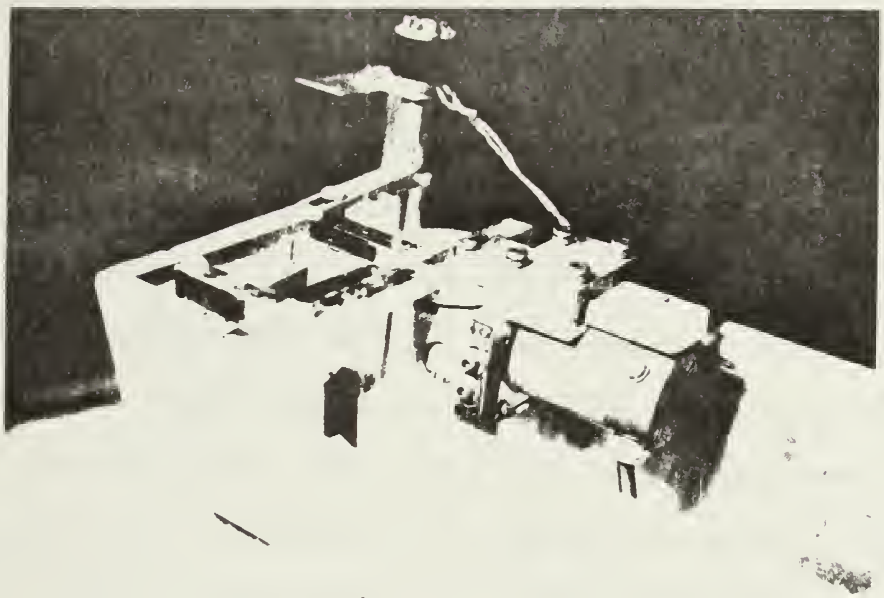
A rotary transducer was used to record the rudder angle. The transducer can be mounted on a calibration stand at the MIT Towing Tank. The location of the zero position of the transducer was found, and the transducer was calibrated from fixed angles marked on the calibration stand.

The accelerometer was calibrated by laying it on a flat horizontal plate. In that position there was a zero reading. Then the accelerometer was turned on its side and measured the acceleration of gravity to calibrate the strip chart recorder.

Once the above calibration procedures were completed, the instruments were again mounted in the ship model in the locations used in the model ballasting procedure. The rotary transducer was mounted on the port rudder stock as shown in Figure 7. The accelerometer was mounted on the ship centerline on a flat metal plate beneath the propulsion

FIGURE 7

ROTARY TRANSDUCER



motor. The rate gyroscope was mounted on a wooden plate located near Station 5.

The measurement instruments were connected to the power supplies and strip chart recorder through approximately 80 feet of cable. The propulsion motor and scotch yoke were connected to power supplies through the same cabling system. Schematics of the electrical connections are shown in Figure 8.

The bandpass filters shown in the schematic of Figure 8 were used after some initial tests were made with the model. A record of the initial data is seen in Figure 9. As would be expected from such measurements, there is a great deal of high frequency noise from machinery vibrations. It can be seen from the roll rate measurement that the matural roll frequency of the model is easily excited and is of such a magnitude as to completely mask out any roll of the same frequency as the rudder oscillations. Since the area of interest in these experiments is only in the frequency of the rudder oscillations, the bandpass filters were used to filter out all but the frequency of interest. The filters were set with a pass band from 0.02 to 0.20 cycles per second. Appendix 4 is a listing of the various switch and dial settings on the bandpass filters during the model tests.

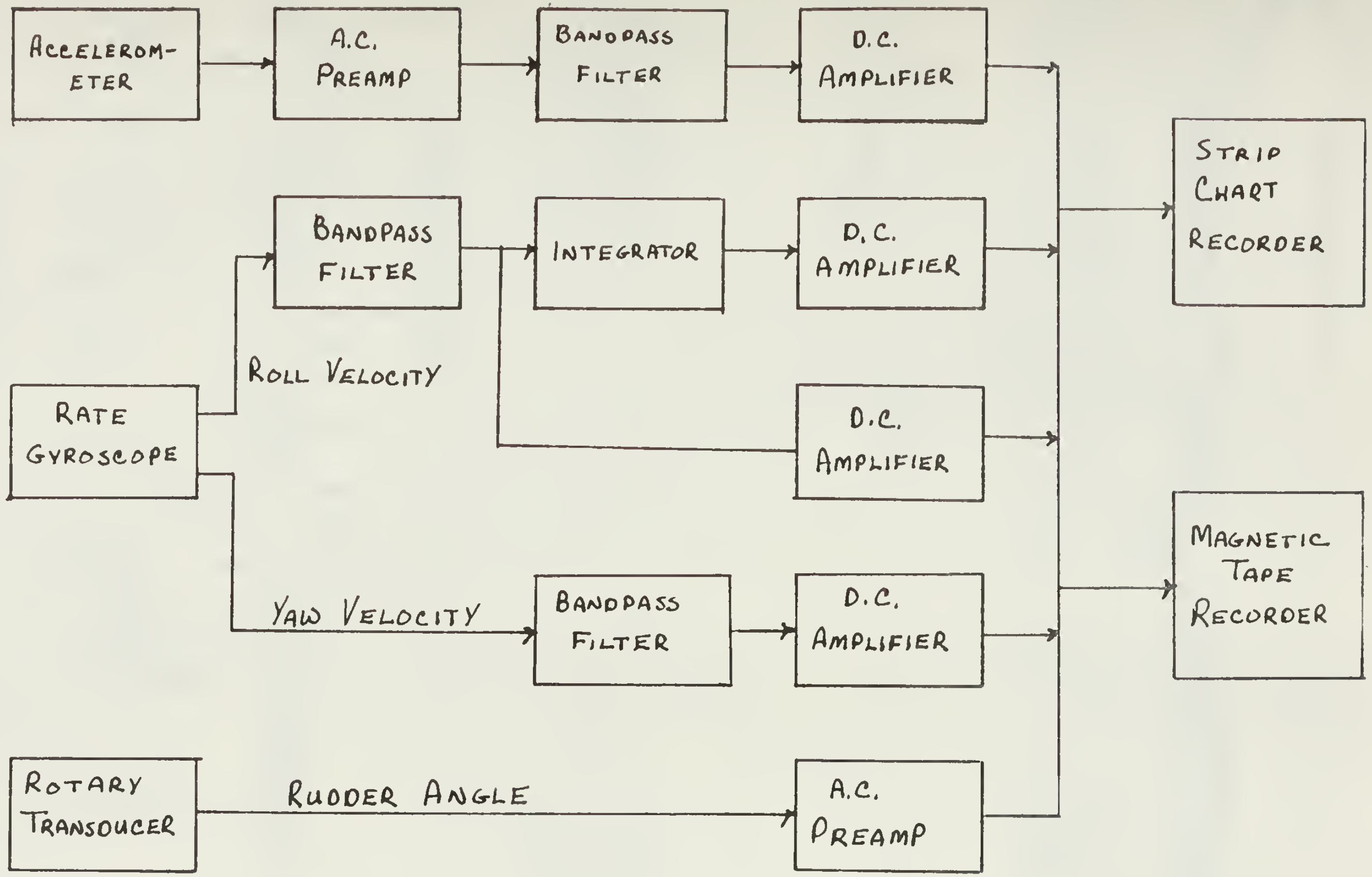
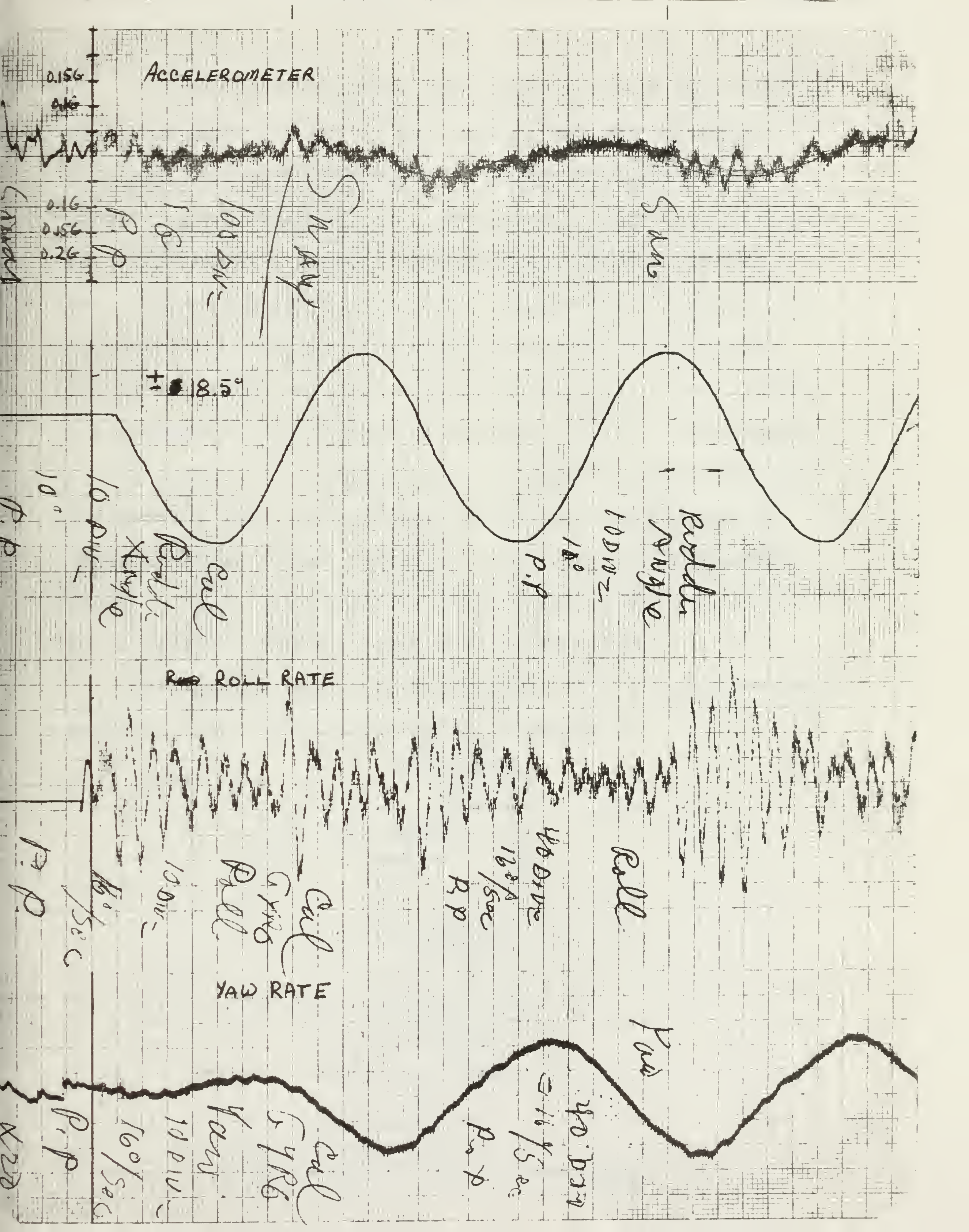


FIGURE 8

SCHEMATIC OF INSTRUMENT CONNECTIONS

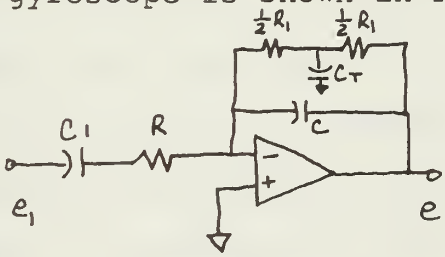
FIGURE 9

ORIGINAL DATA



The use of the filters also necessitated a modification of the calibration of the rate gyro. The output signal from the rate gyro is a DC signal; when the gyro is rotating at a constant rate. When such a signal is applied to a very heavily damped circuit as the bandpass filter, the response of the filter is not adequate for calibration. To remedy the problem, the calibration was first carried out as described earlier without the filter in the circuit. A sinusoidal signal generator was then set at the frequency of the rudder oscillations, and its output voltage adjusted to give the same peak voltage as the rate gyro output at 20 degrees per second rotation. The output of the signal generator was then applied to the circuit with the filter in the circuit and the strip chart calibrated.

The circuit used to integrate the roll rate measurement from the rate gyroscope is shown in Figure 10.



R = 10 MEGAOHM

C = 0.1 μFARAD

R_1 = 20 MEGAOHM

C_1 = 10 μFARAD

C_T = 10μ FARAD

FIGURE 10

HIGH FIDELITY AC. INTEGRATOR

The circuit is taken from Reference 6. The operational amplifier is a Philbrick Model P85AU. Figure 11 shows the circuit gain and phase angle versus frequency for the circuit.

4.3 Ship Model Tests

Once the data from the model tests is recorded on magnetic tape it must be digitized for use with the systems identification program. The digitizing process uses approximately 100 feet of magnetic tape. In order to get enough data at each test speed, it was necessary to complete several runs over the length of the tank. At the beginning of each run the rudders must be placed in the zero position.

The wires for the instruments and power supply to the model were fish-poled into the model. The pole was equipped with two switches to energize and de-energize the propulsion motor and scotch yoke motor.

To perform the tests, an operator must carefully suspend the wires over the model with the fish pole. The propulsion motor is switched on first and the scotch yoke is switched on as the propellers build up thrust and the model moves down the tank. When the model reaches the end of the tank, the motors are switched off, and the operator must tow the model back to the starting position. This

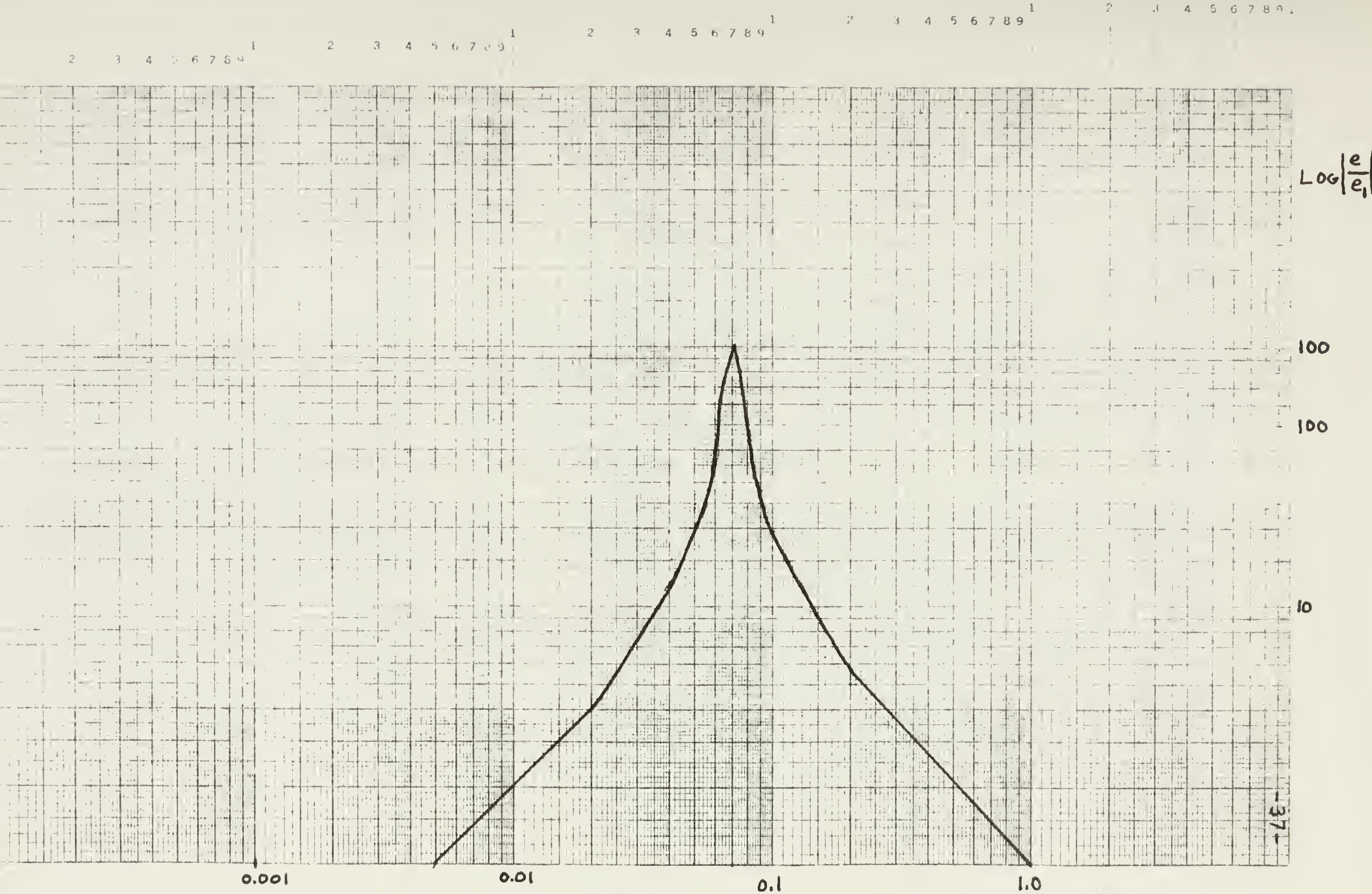


FIGURE 11
INTEGRATOR CIRCUIT GAIN
VS.
FREQUENCY

process is repeated until 100 feet of magnetic tape are obtained for the particular model speed of the test.

CHAPTER 5

DATA ANALYSIS

Samples of the recorded data are shown in Figures 12 and 13. The data was recorded for model speeds of 2.5 feet per second and 1.5 feet per second.

The rate gyroscope used in these experiments has a range of ±500 degrees per second. The gyroscope is linear from 2.5°/second to 490°/second. Thus, the roll rate measurements were not accurately measured in these tests because the rate of roll was less than 2°/second. There also appears to be some random excitations in the roll rate measurement caused by extraneous forces resulting from motions of the wires within the model.

The integrated signal as recorded appears to be the roll angle superimposed on some slow transient response, characteristic of the circuitry and not related to the ship model motions.

The yaw rate measurements from the rate gyroscope were also within the non-linear region of the rate gyro.

For a model speed of 2.5 feet/second, note that the rudder goes through only two cycles before the model reaches the end of the tank. Since both magnitude and phase relationships are important in systems identification, there must be at least two rudder cycles in a particular

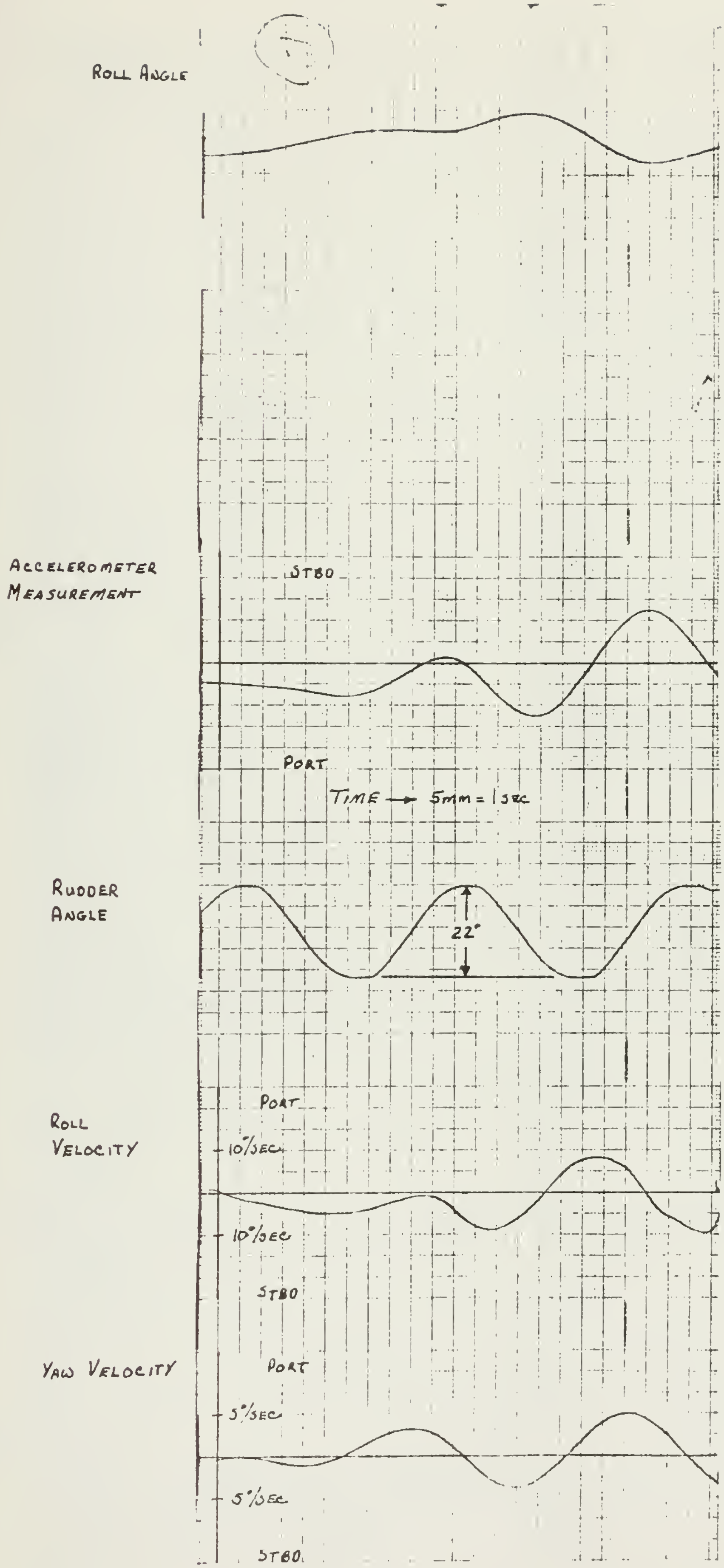


FIGURE 12
RECORDED DATA
MODEL VELOCITY 2.5 $\frac{ft}{sec}$
RUDDER ANGLE ± 11°

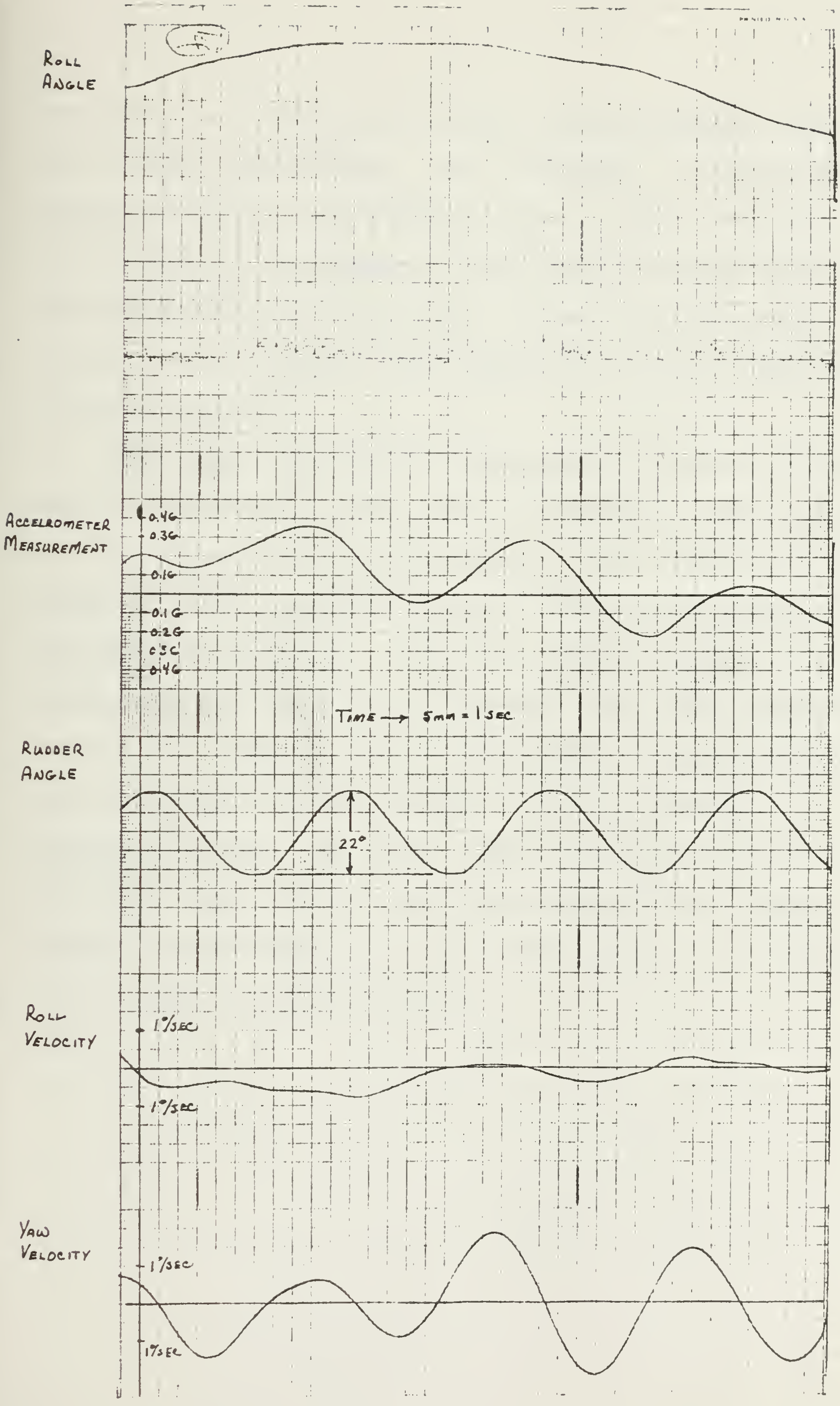


FIGURE 13
RECORDED DATA
MODEL VELOCITY 1.5 FT/SEC
RUDDER ANGLE ±11°

model run to clearly see the phase relationships between the various measurements. For this model 2.6 feet/second scales up to 13.6 knots for the ship. Hence, there are limitations on the speeds at which models can be tested in the MIT tank by this technique. This is not a serious limitation since again the interest is in the linear region of model motions.

Peak values of all the measurements increase as the model moves down the tank. This is expected since rudder lift is proportional to the square of the velocity, and it takes a finite period of time for the model to reach its steady state velocity. The model speed as measured in these experiments is only an average speed over the complete distance of model travel. At the higher model speed where only two rudder cycles are seen, it is difficult to determine whether the steady state condition has been reached. This is a very important consideration since systems identification assumes a condition of equilibrium of a constant forward velocity.

CHAPTER 6

RECOMMENDATIONS AND CONCLUSIONS

6.1 Ship Model Improvements

Several improvements could be made in the self-propelled ship model to better suit it for these tests. Reduction in weight of individual components, elimination of wires fish-poled into the model, and reduction of noise and vibration in the model are those features which will significantly improve the test procedure and data recorded.

In the tests performed for this thesis, the scaled down ship weight was exceeded in the ship model even before ballasting. The components of the maneuvering system and propulsion system are located aft of the ship mid-section which can result in the addition of large amounts of ballast exceedingly far forward of the midship section to reduce trim by the stern. The propulsion motor used in these tests weighed approximately seven pounds, and its one-twentieth horsepower is much more than is necessary to propell the 75-inch model down the tank. Several less powerful and lighter weight motors are available at the MIT Towing Tank and the feasibility of using those motors in any further tests should be investigated. The scotch yoke device is a relatively lightweight component, and the availability of any alternatives may be limited.

Stiffness of the wires leading into the ship model can severely affect the ship model because of possible spurious sway forces and roll frequencies transmitted to the model. The magnitude of the effect of the stiffness of the wires is somewhat dependent upon the type of model under test. For the destroyer model used in these tests, the effects would appear to be more profound than for ship types such as the Mariner used in Reference 3 because of the lighter weight and smaller metacentric height. The ability to eliminate these problems are dependent upon the resources of the experimenter. A radio-controlled model with completely self-contained instrument and data recording devices would obviously completely eliminate the problems. Somewhat more modest solutions could significantly reduce the effect of the wires. The wires to the propulsion motor and scotch yoke are much stiffer than any others leading into the model. Combined with a less powerful propulsion motor, a small battery power supply within the model could get rid of the two stiffest wires. About two hours is all the time necessary to record approximately one hundred feet of data on magnetic tape. Much of that time is taken up in towing the model back down the tank to the starting position and preparing for another run. So the battery endurance could possibly be on the order of an hour long.

Other wires to the measurement devices in the model are not so stiff as those wires leading to the propulsion motor and scotch yoke. Their effect on ship model motions is tolerable. For destroyer type ship models, as used in these tests, any wires into the model can excite the natural roll frequency of the model, and the magnitude of those motions are so great as to prevent accurate recording of the desired motions which are of a lesser magnitude. Where the frequency of the rudder oscillations is much different from the natural roll frequency of the model, a bandpass filter is an adequate solution.

Noise and vibration effects within the ship model can be filtered out of the measurements as well as the roll effects mentioned above. However, noise and vibration can be indicative of poor machinery alignment which can damage the model and impair testing. The thin fiberglass hull of the model used in these tests is a very poor foundation for motor mounts and shaft supports. It is also difficult to accurately position components such as rudders and propellers relative to the model centerline, waterline, or midsection. These problems are probably obvious to those experienced in model testing procedures. Such problems as machinery alignment and mounting should be addressed early in the model design.

One further feature deserves some attention. That is the scotch yoke mechanism. To get a purely sinusoidal motion from the device, care must be exercised to ensure that there is very little play in the various mechanical linkages between turntable, linear motion arm, and tillers.

6.2 Testing Procedure and Test Equipment

To get accurate information for use with systems identification, there are several improvements in test procedure and test equipment that will yield usable data. A different rate gyroscope is needed. The measurement of roll must still be accomplished either by integration of the roll velocity or direct measurement of roll angle. Experiments must be carefully designed to keep the motions linear. The model speed must be low enough to allow several rudder cycles after the steady state velocity is reached, or the time that the steady state velocity is reached must be accurately measured.

The rate gyroscope should operate over a much smaller range of about ±20°/second. For the tests performed on the Mariner model of Reference 3, the yaw velocity was on the order of 10°/second with rudder motions in excess of ±30 degrees. On the destroyer model where the rudder motions were limited to ±11 degrees, the yaw velocity and roll

velocity measurements were on the order of 5°/second or less, which is not in the linear region of the ±500°/second range of the rate gyroscope used.

There are two possibilities for measurement of the model roll angle. These are integration of the roll velocity as attempted in this experiment, or installation of another rotary transducer in the model to measure roll. The latter solution has certain disadvantages in that it would be necessary to add another wire to those already in the model. The addition of the rotary transducer to measure roll angle would also add to the weight and equipment mounting problems encountered in this experiment. Designing an integrator to work properly with other test equipment and at the proper frequency would seem to be the better solution. This is particularly so since it is likely that any rate gyroscope used to measure yaw velocity will be of the three axis types; thus eliminating the need for more wires and equipment in the model.

As mentioned earlier, the model speed measured in these experiments is an average speed over the length of the tank. The average speed is measured by timing, with a stopwatch, the model as it moves down the tank. With the installation of battery power into the model, as suggested earlier, it will be necessary to install a device to count shaft

rotations to ensure constant propeller thrust and thus constant model velocity. The time required for the model to reach a steady state speed is important. The tests should be conducted at a model speed that allows two or three rudder cycles after the model reaches steady state velocity and before reaching the end of the tank.

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APPENDIX 1

LIST OF EQUIPMENT

Power Supply

Propulsion Motor:	Minarik Electric Company Los Angeles, CA Model W33 For Bodine NSH-34 Input 115V - 50/60 Hz 2 Amperes
Rudder Motor:	NJE Corporation Kenilworth, NJ Model TR 36-2 Output 0-36V DC, 0-2 amp

Electric Motors

Propulsion Motor:	Bodine Electric Company Volts 115V DC, 0.65 A RPM 1725 1/20 HP Continuous Duty Weight 7.405 lbs.
Scotch Yoke:	Universal Electric Company Model 7-039 115V DC, 0.34 amps RPM 23,000

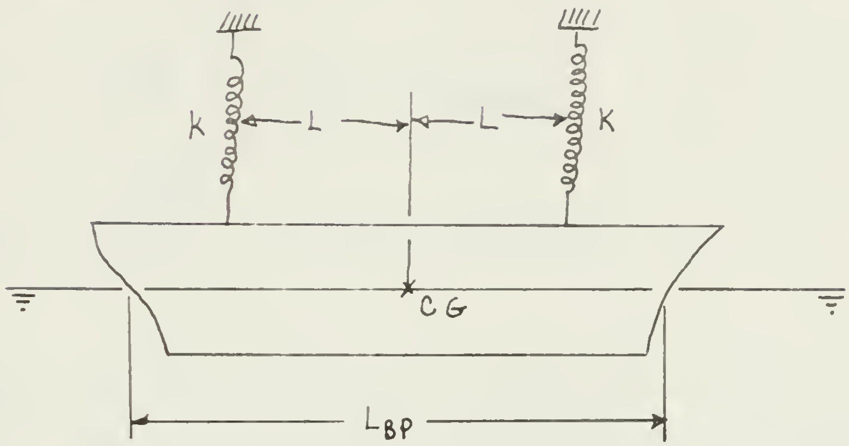
APPENDIX 2

LIST OF TEST EQUIPMENT

Rate Gyroscope:	Northrup Nortronics MFRS PT NO 79183-302 Ser. No. 1 Weight 1.8 lbs. 3-axis gyroscope Range A Axis - ±500°/second B Axis - ±500°/second C Axis - ±500°/second Input Power ±28V DC, 13 Watts maximum Linearity: 1/2% of full scale from zero to one-half scale 2% full scale from one-half scale to full scale
Accelerometer:	Stathem Instruments A6a 1.5-350 Ser. No. 3023 9V maximum, ±1.5G
Rotary Transducer:	Information not available.
Power Supply:	Harrison 6443B DC Power Supply Hewlett-Packard 0-120V, 0-2.5A
Chart Recorder:	Sanborn 7700 Series Recorder
Tape Recorder:	Sanborn 2000 Series Recorder
Cables:	2 - Belden 5-conductor shielded cable 1 - Belden 8254 RG 62/U gyro 2 - US Wire 92194 RG 174/U gyro

APPENDIX 3

MODEL BALLASTING PROCEDURE



Theory: For an elongated body the radius of gyration in pitch approximates that in yaw. The moment of inertia J of the ship is dependent upon the mass of the ship, m, and the radius of gyration R.

$$J = mR^2$$

For most ships the relation between the radius of gyration and the length between perpendiculars, L_{BP} is

$$\frac{R}{L_{BP}} \cong 0.25.$$

For the model suspended as shown in the above figure the pitch and heave frequencies are

$$\omega_{HEAVE} = \sqrt{\frac{2K}{m}} = \text{heave frequency,}$$

$$\omega_{PITCH} = \sqrt{\frac{2KL^2}{J}} = \text{pitch frequency.}$$

Now from the relationships for moment of inertia and the ratio of radius of gyration to L_{BP} one can determine the moment of inertia.

$$J = m\,(0.25)^2\,L_{BP}^2$$

$$\frac{\omega_H}{\omega_P} = \frac{\sqrt{\frac{2K}{m}}}{\sqrt{\frac{2KL^2}{J}}} = \frac{1}{\sqrt{\frac{L^2}{J/m}}} = \frac{1}{\sqrt{\frac{L^2}{(0.25)^2 L_{BP}^2}}}$$

$$\therefore\ \frac{\omega_H}{\omega_P} = 0.25\ \frac{L_{BP}}{L}$$

Procedure

1. Make sure the ship model, without ballast, rests on a level keel, or a desired trim, when put in water, and determine the center of gravity of the model. Assume that the center of gravity coincides with the center of flotation. Now put all ballast in the model symmetrically about the C.G.
2. Suspend the two springs from convenient points on the ceiling. The distance between the two springs should be 2L.
3. Measure off the distance L on each side of the center of gravity of the ship model. L can be any convenient length.
4. Suspend the model from the springs, using string wrapped around the model and secured in place by tape as shown in the figure above.
5. Force the model to an up and down motion (heave) and count with a stopwatch the number of oscillations per minute. Thus, determine f_H.
6. Force the model to a pitching motion and count with the stopwatch the number of oscillations per minute. Thus, determine f_p. It is not necessary to achieve a pure pitching motion, completely uncoupled from

heave, because the period of heave does not affect the period of pitch.

7. Now determine f_H/f_p and $0.25\ L_{BP}/L$. The heave frequency f_H remains constant for ω given model weight.

If $f_H/f_p > 0.25\ L_{BP}/L$, f_p is smaller than required. To increase f_p, dispose ballast weights closer to the C.G., making sure it is still symmetrically disposed about the C.G. Then repeat step 6 and check the new ratio f_H/f_p.

If $f_H/f_p < 0.25\ L_{BP}/L$, f_p is larger than required and the ballast weights should be moved further away from the C.G.

When $f_H/f_p = 0.25\ L_{BP}/L$ the ballasting procedure of the ship model is over.

APPENDIX 4

SWITCH AND DIAL SETTINGS

FOR

BANDPASS FILTERS

Low Cutoff Frequency	0.02 Hz.
High Cutoff Frequency	0.20 Hz.
Input Amplitude Switch	Low
Peaking Switch	Standard

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